Proceedings of DETC2003
2003 ASME Design Engineering Technical Conferences
September 2-6, 2003, Chicago, Illinois, USA

DETC2003/PTG-48082

UNDERSTANDING VIBRATION SPECTRA OF PLANETARY GEAR SYSTEMS FOR FAULT DETECTION

Marianne Mosher
NASA Ames Research Center
M.S. 269-3
Moffett Field, CA 94035-1000
650-604-4055, Marianne.Mosher@nasa.gov

ABSTRACT

This paper explores the vibration spectra for planetary gear systems by studying a kinematic model of vibration and comparing the model with measurements of two helicopter transmissions made in flight. The model and flight data include systems with both uniformly and nonuniformly spaced planet gears. This model predicts vibration to occur only at frequencies that are integer multiples of the planet spacing repetition frequency and clustered around gear mesh harmonics. Vibration measurements show the model correctly predicts the frequencies with large components around the first several harmonics of the gear mesh frequency. Measurements do not confirm some of the more detailed features predicted by the model. Some features in the spectra from the numerically derived model can be used to separate the model data with and without planted faults. These features were not found useful for detecting faults in the vibration measurements of real gearboxes in flight due to added complexity in the spectra from real gearboxes.

Keywords: *Flight test, Helicopter transmission, Kinematic model, Planetary gear, Vibration measurement.,*

INTRODUCTION

An understanding of vibration spectra is very useful for any gear fault detection scheme based upon vibration measurements. The vibration measured from planetary gears is complicated. In this paper, the term planetary gear system refers to the compound gear systems with planet gears between a center sun gear and an outer ring gear, with the ring gear fixed and not rotating. Planetary gear systems (Fig. 1) provide coaxial gear reductions and are useful for machinery with high power requirements such as helicopter transmissions. Sternfeld [1] noted the presence of sidebands about the gear mesh harmonics spaced at the planet passage frequency in spectra measured near the ring gear of a CH-47 helicopter. McFadden [2] proposed a model of the vibration transmission that predicts high spectral amplitudes at multiples of the planet passage frequency (number of planets times planet carrier revolution frequency), for planetary gears with evenly spaced planets. This model correctly predicts shifting of the strong signal from a gear mesh frequency to a sideband of the meshing frequency when the number of teeth on the ring gear is not an integer multiple of the number of planets.

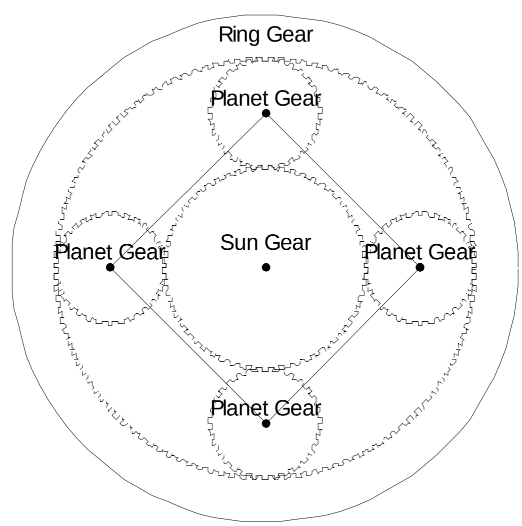

Fig. 1 Planetary Gear System

The spectra of vibration measured from planetary gear systems are more complicated than the spectra measured from simple gear mesh pairs. For simple gear mesh pairs in normal operation, most of the vibration occurs at the gear mesh frequency and it's integer harmonics. For a planetary gear system, most vibration measured in the reference frame of the transmission occurs at various sidebands of the gear mesh frequency and its integer harmonics. This added complexity in planetary gear systems invalidates the use of the various metrics developed [3-6] to test for faults in gear pairs. Planetary gear vibration signal separation schemes have been developed with reported success by McFadden [7-9], Forrester [10] and Samuel [11] to enable the detection of faults in planetary gear systems. For separation, the signal attributed to each planet gear is assembled from parts of the measured signal when the planet gear is closest to the measurement transducer. These signal separation methods all require knowing the location in time when a planet gear passes closest to the measuring accelerometer. Planet passage detection by use of a carrier phase signal requires the use and maintenance of the reference phase angle. Determining the planet passage from the vibration signal itself can be more difficult than one might expect. The separation can become more difficult with larger numbers of planet gears. Unevenly spaced planet gears add more complexity to the signal separation task.

This study examines the spectra of planetary gear vibration by means of a model and flight measurements. The goal is to gain a better understanding of these spectra, which could lead to the discovery of features in the spectra that could be utilized in fault detection. The model and the analysis of flight measurements will be described. Spectra will be shown from numerically derived model data with and without added noise, tooth-to-tooth variation and planted flaw. Spectra will be shown from measurements made of two four-planet systems, one with uniformly spaced planet gears the other with nonuniformly spaced planet gears. The signal of the numerically derived model can be made to approximate the signal of the measured helicopter system by setting model parameters judiciously.

NOMENCLATURE

C	Parameter, sum of complex exponential functions
$\mathbf{G}$	Fourier coefficients for a planet gear mesh signal
J	Number of planet gears
N	Number of teeth on ring gear
R	Repetition frequency of planet passage spacing
T	Time for one planet carrier rotation
$\mathbf{V}$	Fourier coefficients for vibration signal from a single planet gear at measurement location
$\mathbf{W}$	Fourier coefficients for window function representing the amplitude variation of the vibration from a planet gear as the gear revolves around the sun gear
Z	Set of all integers
g	Time domain vibration from a planet gear mesh
i	$\sqrt{-1}$
j	Integer index for planet
k , l	Integer index for Fourier coefficient
m	Integer index
q	Time delay of planet passage
t	Time
v	Time history vibration signal from a planetary gear system at measurement location
w	Window function representing the amplitude variation of the vibration from a planet gear as the gear revolves around the sun gear

MODEL DEVELOPMENT

A model of an idealized signal from a planetary gear system is developed. A planetary gear system contains a central sun gear, a coaxial outer ring gear and several planet gears between the sun and ring gears. The sun and ring gears can rotate. The planet gears rotate and can revolve around the sun gear while connected to each other through a carrier cage. One of the planetary gear elements, ring gear, sun gear or planet gears, remains stationary while the other two move around the center. This study concerns the case with a stationary ring gear. Both planet/sun and planet/ring meshings produce vibrations.

Assume that the planet gears are identical. Also, assume that the teeth in the sun gear are identical to each other, the teeth in the ring gear are identical to each other and the teeth on the planet gears are identical to each other. The uniformity of gear teeth leads to each planet/ring tooth interaction being identical and producing identical local response in the structure; and the same is true for each planet/sun tooth interaction. One way to measure vibration is with an accelerometer. An accelerometer will measure the gear mesh signals from all of the interactions after they propagate through the structure. With J planets, the total signal at the accelerometer may be modeled as the sum of the signals from each planet gear with the signal from each planet gear represented as the product of a window function, w_j, modeling the amplitude of the transfer function from the gear vibration source times a gear mesh function, g_j, modeling the gear vibration source. For planetary gear systems in helicopters, both the amplitude and time derivative of the retarded time delay are insignificant. The wavelength of the first harmonic is of the order of 10 m, much larger than the difference in distance from the transducer to the near and far ranges of the planet mesh vibration source. The Mach number of the vibration source is of the order of .001 to .01. Since both time delay wave propagation effects and Doppler effects are negligible, the retarded time delay will be considered zero in this model, thus ignoring wave propagation effects to make a much simpler model. The model equation with q_j as the delay time associated with each planet revolving around the center of the system is:

$$v(t) = \sum_{j=1}^{j=J} w_j\left(t - q_j\right) g_j\left(t - q_j\right) \qquad (1)$$

Transforming Eq. (1) to the frequency domain with a Fourier series yields:

$$\mathrm{V}[k] = \frac{1}{T}\sum_{j=1}^{J}\sum_{l=-\infty}^{\infty} \mathrm{W}_j[k-l]\,\mathrm{G}_j[l] \qquad (2)$$

$$\mathrm{W}_j[k] = \mathrm{W}_1[k]e^{-2\pi ik(q_j - q_1)/T}$$
$$\mathrm{G}_j[k] = \mathrm{G}_1[k]e^{-2\pi ik(q_j - q_1)/T} \tag{3}$$

Define a parameter $C(k)$ equal to the sum of the exponential terms

$$C(k) = \sum_{j=1}^{j=J} e^{-2\pi ik(q_j - q_1)/T} \tag{4}$$

The gear mesh function, g, is periodic with the gear mesh frequency, so the coefficients of its Fourier series have non-zero values only at integer multiples of the gear mesh frequency. For the period of one planet gear revolution, non-zero values occur only at integer multiples of N, the number of teeth on the ring gear,

$$\mathrm{G}_1[l] = \begin{cases} \mathrm{G}_1[l], \; l = 0, \pm N, \pm 2N, \ldots \\ 0, \text{ otherwise} \end{cases} \tag{5}$$

Combining Eqs. (2), (3), (4) and (5) leads to:

$$\mathrm{V}[k] = \frac{C(k)}{T} \sum_{m=-\infty}^{m=\infty} \mathrm{W}_1[k - mN]\mathrm{G}_1[mN] \tag{6}$$

Now assume that the window function contains most of its energy at relatively low frequency so that its spectrum is significantly decayed and stays decayed at harmonics indexed above one half the number of teeth on the ring gear. For such well-behaved windows, one term in Eq. (6) will dominate the series. The dominant term is the term that minimizes the difference in the index of the gear mesh frequency with the index of the complete planetary gear signal. So choose m_k so that $|k - m_k N|$ is a minimum, then the series may be approximated by its dominant term,

$$\mathrm{V}[k] \approx \frac{C(k)}{T} \mathrm{W}_1[k - m_k N]\mathrm{G}_1[m_k N] \tag{7}$$

For example, if the ring gear contains N teeth, the terms in the window function that contribute to a term in the total signal will be spaced at the interval of N. Unless the window function is very spiky, only Fourier coefficients with indices much less than N will have significant amplitude. Thus, at most, only one term in Eq. (6) will be significant. Moreover, for values of the index k far from any gear mesh harmonic, all the terms are insignificant compared to the energy in the total signal.

For evenly spaced planet gears, the time offset between them are equally spaced through the time, T, of a planet revolution. With this simplification, the parameter $C(k)$ takes on a simple form,

$$C(k) = \begin{cases} J, \; \dfrac{k}{J} \in Z \\ 0, \text{ otherwise} \end{cases} \tag{8}$$

because the individual complex exponential terms take on the value of one when k/J is an integer since $k(q_j - q_1)/T = k(j-1)/J$ is then always an integer. When k/J is not an integer, the complex exponential terms are evenly spaced around the unit circle and sum to zero.

For systems with unevenly spaced planet gears, the ratio of the frequency index to the repetition frequency, k/R, is the relevant parameter for determining when the sum $C(k)$ is zero or nonzero. The repetition frequency, R, is the frequency at which the planet passage spacing pattern repeats itself:

$$C(k) \neq 0, \frac{k}{R} \in Z$$
$$C(k) = 0, \text{otherwise} \tag{9}$$

When analyzed with a period equal to the planet revolution, the Fourier Series for the model of the idealized signal from a planetary gear with evenly spaced planet gears contains nonzero terms only for frequencies that are integer multiples of the number of planets in the gear multiplied by the planet revolution frequency. The energy is clustered in sidebands about the integer multiples of the gear mesh frequency. If the number of teeth in the ring gear is not an integer multiple of the number of planets, there is no signal at the gear mesh frequency according to this analysis. A more detailed development of this mode will be published at a later date.

With this simple kinematic model of the vibration signal from a planetary gear system, some features of the signal are predicted without knowledge of the actual forcing function or transfer function. The model predicts the frequencies where larger amplitudes are expected using only the tooth count of the ring gear, number of planet gears and relative spacing of the planet gears. Vibration energy from this ideal model will occur only in clusters of side bands and some gear mesh harmonics around the harmonics of the gear mesh frequency. Only side band and harmonic frequencies that are integral multiples of the repetition frequency will contain vibration energy. In a system with evenly spaced planet gears, the repetition frequency is the planet passage frequency. In a system with non-uniformly spaced planet gears, the repetition frequency occurs between the carrier frequency and the planet passage frequency. For example, the transmission in the OH-58C contains four planet gears spaced in pairs that are 180 deg apart and intersect at about 91.4 deg. In this system, the repetition frequency is twice the carrier frequency since the planets are 180 deg. This simple kinematic model explains why the amplitude of pure gear mesh frequency is in the noise level instead of high for some planetary gear systems; vibration energy measured in the fixed reference system only occurs at multiples of the repetition frequency. When the gear mesh frequency is not a multiple of the repetition frequency there will be no energy at the gear mesh frequency when analyzed with a block size equal to an integer number of carrier rotations.

Some assumptions in this idealized model are unrealistic for real planetary gear systems. Even the most precisely manufactured gear system contains tooth-to-tooth variations in its geometry and material properties leading to tooth-to-tooth variation in the vibration forcing function. Noise is always present from various sources. To consider these effects, noise will be added to the model signal and amplitude modulation will be applied to the "ideal" gear mesh vibration signals. To consider the effect of a gear flaw, phase modulation will be applied to a short section of an ideal planet gear mesh signal.

ANALYSIS OF FLIGHT MEASURMENTS

NASA Ames Research Center has been measuring vibration of helicopter transmissions in flight tests since 1998. Ames' researchers tested the AH-1S helicopter in 1998 and 1999 and the OH-58C helicopter in 2000 in a series of controlled flight conditions throughout the flight envelope, see Huff [12, 13] for details. The AH-1S helicopter has a gross weight of 4000 kg and its transmission contains two planetary gear systems. The ring gear contains 119 teeth, the sun gear contains 57 teeth and the planet gears contain 31 teeth in both systems. The upper planetary system contains eight uniformly spaced planet gears and the lower planetary system contains four uniformly spaced planet gears. The OH-58C has a gross weight of 1280 kg and contains one planetary gear system with 99 teeth on the ring gear, 27 teeth on the sun gear and 35 teeth on the planet gears. This system contains four non-uniformly spaced planet gears; the planet gears come in pairs spaced 180 deg apart with the angle between the pairs other than 90 deg.

The AH-1S was instrumented with two tri-axial accelerometers on the transmission cover, one near the upper planetary ring gear and the other near the lower planetary ring gear. The OH-58C was instrumented with one tri-axial accelerometer and three single axis accelerometers, all mounted on the casing around the ring gear. On both helicopters, torque was measured by calibrating the oil pressure and the main rotor shaft was instrumented with a 1/rev signal generator. Vibration data, oil pressure for torque and a 1/rev signal were collected with a pc-based digitized system on board the aircraft. The anti-aliasing filter was set to 18 kHz and sample rate was 50 kHz.

For this study, the time histories of all vibration measurements were interpolated to a fixed number of samples per rotation of the carrier cage of the planet gear. Vibration data from the AH-1S upper planetary gear and OH-58C were interpolated to 8192 samples per rotation. Data from the AH-1S lower planetary gear were interpolated to 4096 samples per rotation. All of these resampling frequencies are consistent with the Nyquist criteria. Amplitude spectra were made from time synchronous averages of the signals and amplitude spectra were also made from averaging the power spectra of individual blocks of data one carrier rotation long. With the time synchronous averaging, frequency components that are not integer multiples of the carrier rotation frequency will be reduced in amplitude and thus the periodic componant of the planetary gear vibration will be emphasized. The amplitudes of both random frequency components and discrete frequency components not commensurate with the carrier frequency will be reduced. On the other hand, with averaging power spectra, all frequency components are retained whether periodic with the carrier rotation or not. All Fast Fourier Transforms were made with a rectangular window on the data blocks. The rectangular window has a benefit for the frequency analysis of data when the frequencies of interest are exact integer multiples of the inverse of the block size, as was set up with the interpolation to a fixed number of sample points per carrier revolution. Under these specific circumstances, the benefits of the rectangular window include the conservation of energy in the transformed signal, no alteration of amplitude of periodic frequency components due to window effects and the smoothing of non-periodic components in the frequency domain.

RESULTS FROM MODELING

Numerically generated planetary gear vibration data were produced based upon the ideal model described above. Spectra were calculated with and without noise, amplitude modulation on the gear mesh signal and phase distortion on a section of a planet gear. A system was chosen with 119 teeth on the ring gear and 4 equally spaced planet gears with 31 teeth. The block size in the spectrum is one carrier rotation, thus frequency index 119 is the gear mesh frequency and frequency index 4 is the repetition frequency. Figure 2 shows the spectrum for the ideal case with significant amplitudes only at multiples of 4.

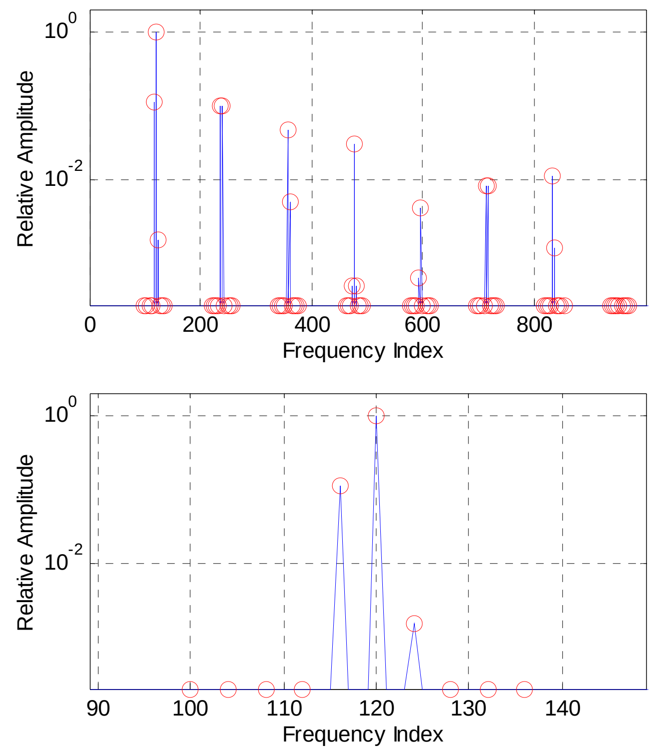

Fig. 2 Spectrum for ideal model planetary gear vibration signal

Circles indicate frequencies where the ideal model predicts non-zero amplitudes. Large amplitudes occur at frequency indices 116, 120, 236, 240, 356, 360, 476, 592, 596, … and so on. The form of Eq. (7) predicts certain patterns and symmetries in the amplitudes of these frequencies by the form. Only a small number of the low frequency window coefficients contribute any significant energy. One of these low frequency window components and its equal amplitude negative frequency componant will modulate more than one of the gear mesh harmonics. In this example with four uniformly spaced planet gears, the relative amplitude of the sidebands to the forcing gear mesh harmonic will repeat every forth gear mesh harmonic. For example the relative amplitudes around the first gear mesh harmonic, 119, have the same pattern as the relative amplitudes around the fifth gear mesh harmonic, 595. In addition, the relative amplitudes of the sidebands around 357, the third gear mesh harmonic, are identical to the relative amplitudes of the sidebands around the first and fifth gear mesh harmonics in reverse order. The final pattern in this example is that the sidebands around all of the even numbered gear mesh harmonics have symmetric amplitudes.

Figure 3 shows the spectrum when noise is added to the example model. In this case, the noise raises the level of the zero and low amplitude frequency components and slightly modifies the amplitude of the significant components as can be seen in the sidebands of the sixth gear mesh harmonics which are no longer symmetric.

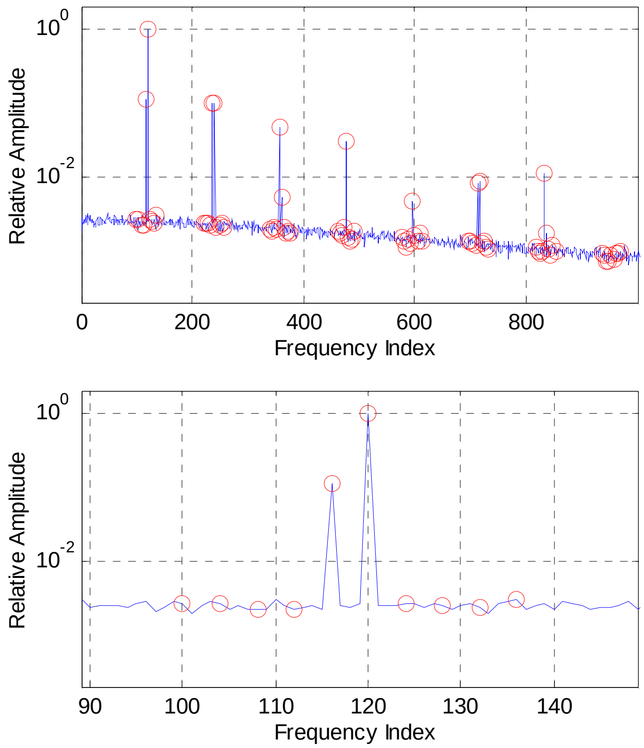

Fig. 3 Spectrum for ideal model planetary gear vibration signal plus noise

Figure 4 displays the spectrum with the original planet gear signal modified so that the amplitude varies for each tooth in each planet gear and a smaller variation is added to each rotation of the planet gears. In comparison with the ideal model (Fig. 1), more broadband energy appears between the gear mesh harmonics and there appears to be more energy and feature variety in the sidebands of the gear mesh harmonics. The vibration signal from a real gear system will have both noise and tooth-to-tooth variations, Fig. 5 displays the model with both these alterations. As before, the addition of noise increases the broadband noise floor.

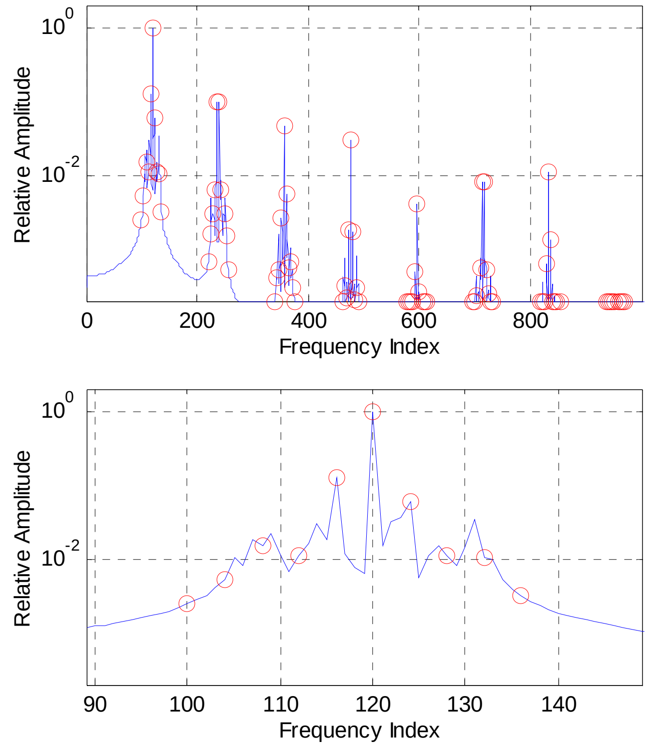

Fig. 4 Spectrum for ideal model planetary gear vibration signal with tooth-to-tooth variation

Next, consider the model with a fault in one planet gear (Fig. 5). This spectrum, where a phase distortion was added to a section of one of the planet gear vibration signals, contains several features not in the spectra from the ideal model nor the ideal model with added noise and tooth-to-tooth amplitude variation. No amplitudes are zero, thus the spectrum looks like more complicated noise has been added. There is a distinct periodic ripple with a frequency near 3, especially visible below the first gear mesh harmonic. In addition, some sculpted enveloping shows. The complex cepstrum of this flawed signal indicates repetitions in the spectrum at about 2.3, 19.7, 29.3 and 56.8 per rotation of the planet carrier. These structures in the spectrum indicate the possibility of devising a test for planetary gear flaws based upon the amplitude spectrum of the vibration signal resampled to a fixed number of samples per carrier rotation. These structures will not be evident unless the time samples are at a constant phase separation instead of constant time separation. However, this particular flaw is extreme; the value of FM4 [3] for the raw gear mesh signal with a flaw and noise is 25.7 while the value of the ideal raw gear mesh signal with noise and without fault is 3.0. FM4 is a metric designed to find anomalies in gear vibration signals. FM4 increases if the signal contains a local increase in amplitude or a local phase distortion. The nominal value of FM4 is 3 and it is not uncommon for values of 6 or lower to be measured in test rigs containing gear faults [5, 14].

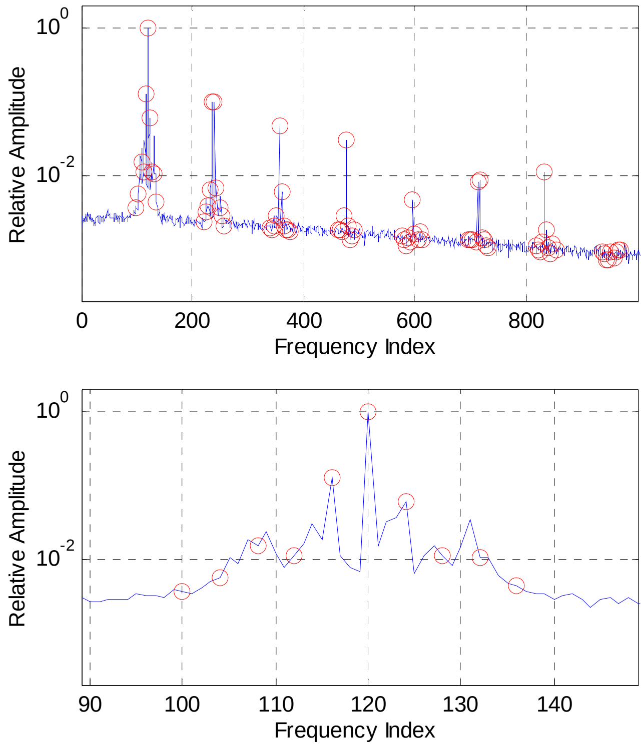

Fig. 5 Spectrum for ideal model planetary gear vibration signal with tooth-to-tooth variation and noise

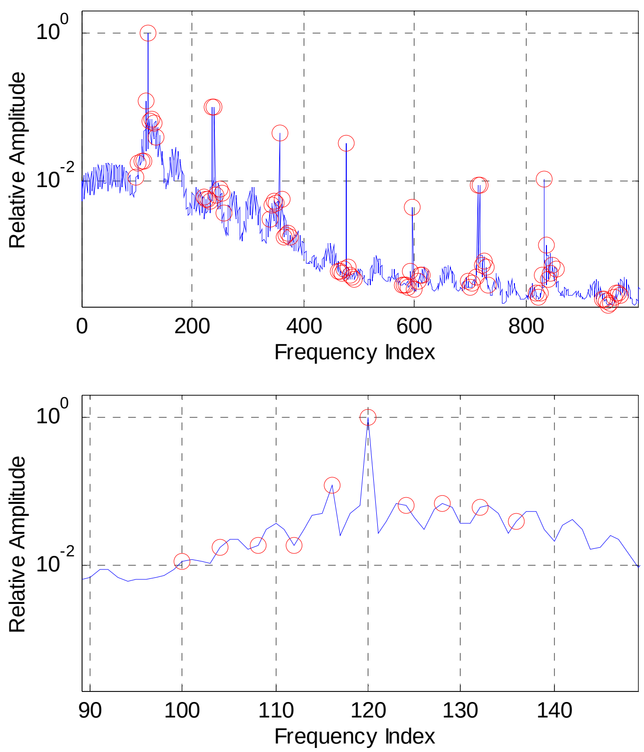

Fig. 6 Spectrum for ideal model planetary gear vibration signal with planted fault

RESULTS FROM FLIGHT MEASUREMENTS

To determine the validity of this ideal model, spectra from flight measurements of helicopters will be viewed and compared with the model. First spectra from time synchronous averages will be viewed; by averaging on the carrier rotation any tooth-to-tooth variations from the planet gear and sun gear will be minimized. The lower planetary gear system on the Cobra has 119 teeth on the ring gear and 4 equally spaced planet gears, each containing 31 teeth, the same gear configuration of the model studied in the previous section. Figure 7 displays a spectrum derived from the time synchronous average of 31 carrier rotations from the lower planet carrier on the Cobra helicopter. Circles indicate the location of frequencies where the model predicts "discrete" components, frequencies that are integer multiples of four and sidebands around 119 and its integer multiples. The high amplitude frequencies identified by the model agree well with the high amplitude frequencies found in the data. The frequencies at the sidebands close to gear mesh harmonics selected by this model are the frequencies near the gear mesh harmonics with high measurement levels, especially at the lower harmonics. The frequency predictions tend not to be exact at the higher order gear mesh, as can be seen around the 8th gear mesh harmonic (frequency index = 952) in this example. Some parts of the spectrum not identified by the model are relatively high, note the region around frequency index 200. The model does not contain all the other vibration sources on the helicopter. The source of the vibration near frequency index 200 comes from some other component not synchronous with the lower planet carrier as can be confirmed by this regions very large relative amplitude in the spectra in Fig. 8 made from a power spectral average instead of a time synchronous average. In the power spectral average, the frequency components identified by the model are still prominent, but not dominant. The relationships between the amplitudes of sidebands predicted by the model are not present in the measured vibration from the flight test; the patterns about the first and fifth gear mesh harmonics differ greatly and the patterns about the even gear mesh harmonics are not symmetric.

The overall image of the measured flight spectra in Figs. 7 resemble the spectra of the model with a planted flaw in Fig. 6 more than any of the models without flaws in Figs. 2-5. There appears to be so much complexity in the vibration signals of helicopter transmissions measured in flight, that the unique features seen in the model containing a simulated fault could easily be masked by complexity in the measured flight data.

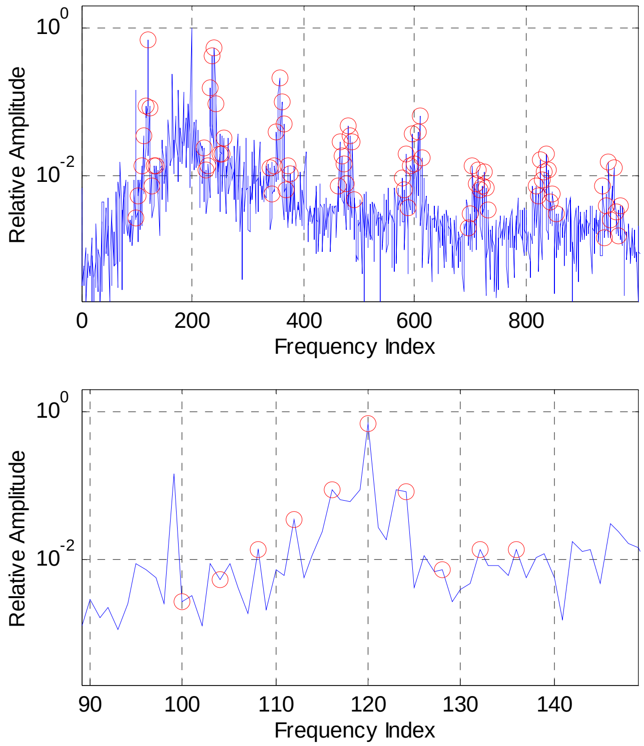

Fig. 7 Spectrum from Lower Cobra Planetary System made from a time synchronous average

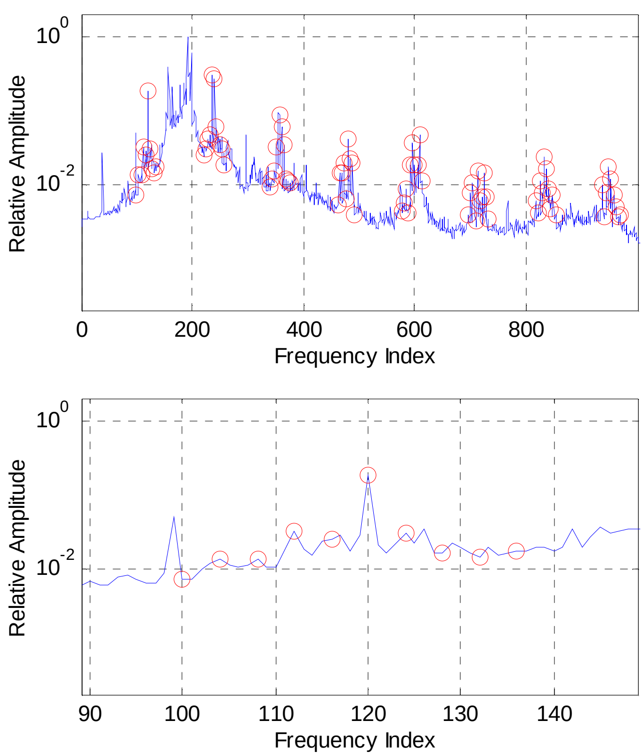

Fig. 8 Spectrum from Lower Cobra Planetary System made from a power spectral average

The ideal model predicts a different set of high amplitude frequencies for a system with unequally spaced planet gears. Ames' researchers measured transmission vibrations on an OH-58C with unequally spaced planet gears. As shown in Fig. 8, high amplitude spectral components occur at frequencies predicted by the model, integer multiples of two at and near integer multiples of the gear mesh frequency, 99. Note in this example, both the discrete frequency components and broadband components are largest near the tenth gear mesh harmonic. This overall spectral shape is believed due to the frequency response of the transducer mounted on a bolt of the transmission housing. In this four planet case, the ideal model predicts all the sideband amplitude patterns of the even gear mesh harmonics will be equal to each other and all the sideband amplitude patterns of the odd numbered gear mesh harmonics will be equal to each other. As with the previous example, the sideband amplitude patterns differ markedly from those predicted by the model. In this example, the model does an excellent job of predicting which frequencies have relatively high amplitudes up through the 11th gear mesh harmonic, and some discrepancies in the 12th harmonic and above even though the broadband noise does not become highly significant until the 15th harmonic.

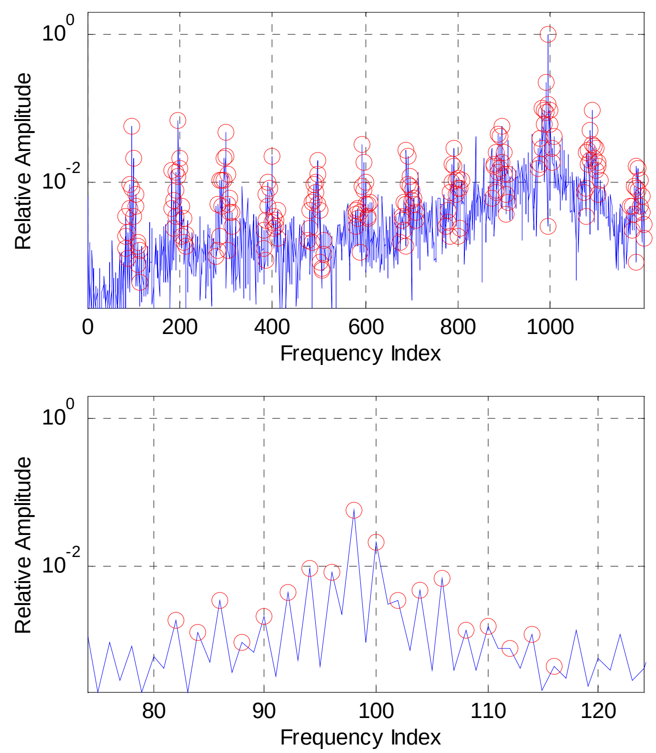

Fig. 8 Spectrum from OH-58C Planetary System made from a time synchronous average

CONCLUSION

A model of idealized planetary gear vibration was developed that explains some characteristics of spectra without specific knowledge of the vibration source or transfer functions. The model predicts discrete frequency components in the spectra at integer multiples of the planet repetition frequency at gear mesh harmonic frequencies and their side bands. The discrete frequencies predicted by the model match the frequencies of large amplitude components in measurements from real helicopter transmissions in flight, especially up to the

10^{th} gear mesh harmonic. Frequencies do not match as well around higher gear mesh harmonics. The model predicts the repetition of relative side band amplitudes and their mirror image at certain related harmonics; measurements of vibration from real planetary gear systems do not show these relations. Some features in the spectra from the numerically derived model can be used to separate the model data with and without planted faults. These features are not expected to be useful for detecting faults in the vibration measurements of real gearboxes in flight due to added complexity in the spectra from real gearboxes.

ACKNOWLEDGMENTS

NASA's Computing, Information and Communications Technology Program funded this work. The author would like to thank the Army Flight Projects Office headed by Major David Arterburn for coordination of aircraft instrumentation and flight support activities. Dr. Eric Barszez provided many useful technical discussions.

REFERENCES

[1] Sternfeld, H., Schairer, J. and Spencer, R., 1972, "Investigation of Helicopter Transmission Noise Reduction by Vibration Absorbers and Damping".

[2] McFadden, P. D. and Smith, J. D., 1985, "An Explanation for the Asymmetry of the Modulation Sidebands about the tooth Meshing Frequency in Epicyclic Gear Vibration", Proceeding of Institute of Mechanical Engineers, **199**, pp.

[3] Stewart, R. M., 1977, "Some useful Data Analysis Techniques for Gearbox Diagnostics", University of Southampton, Southampton, England.

[4] Martin, H. R., 1989, "Statistical Moment Analysis as a Means of Surface Damage Detection", *7th International Modal Analysis Conference*, Society for Experimental Mechanics, Schenectady, NY, pp. 1016-1021.

[5] Zakrajsek, J. J., Towsend, D. P. and Decker, H., 1993, "An Analysis of Gear Fault Detection Methods as Applied to Pitting Fatigue Failure Data", *47th Meeting of Society for Machinery Failure Prevention Technology*, Society for Machinery Failure Prevention Technology, pp. 199-208.

[6] Choy, F. K., Huang, S., Zakrajsek, J. J., Handschuh, R. F. and Townsend, D. P., 1994, "Vibration signature Analysis of a Faulted Gear Transmission System", *30th Joint Propulsion Conference*, AIAA, Indianapolis, Indiana.

[7] McFadden, P. D. and Howard, I. M., 1990, "The Detection of Seeded Faults in an Epicyclic Gearbox by Signal Averaging of the Vibration", Defence Science and Technology Organization Aeronautical Research Laboratory, Melbourne, Victoria.

[8] McFadden, P. D., 1991, "A Technique for Calculating the Time Domain Averages of the Vibration of the Individual Planet Gears and the Sun Gear in an Epicyclic Gearbox", Journal of Sound and Vibration, **144**, pp. 163-172.

[9] McFadden, P. D., 1994, "Window Functions for the Calculation of the Time Domain Averages of the Vibration of the Individual Planet Gears and Sun Gear in an Epicyclic Gearbox", Journal of Vibration and Acoustic, **116**, pp. 179-187.

[10] Forrester, D., 1996, "Method and Apparatus for Performing Selective Signal Averaging", Commonwealth of Australia.

[11] Samuel, P. D. and Pines, D. J., 2000, "Vibration Separation and Diagnostics of Planetary Geartrains", American Helicopter society, Virginia Beach, VA.

[12] Huff, E. M., Barszez, E., Tumer, I. Y., Dzwonczk, M. and McNames, J., 2000, "Experimental analysis of Steady-State Maneuvering Effects on Transmission Vibration Patterns Recorded in an AH-1 Cobra Helicopter", *56th Annual Forum*, American Helicopter Society, Virginia Beach, VA.

[13] Huff, E. M., Tumer, I. Y. and Mosher, M., 2001, "An Experimental Comparison of Transmission Vibration Responses from OH-58 and AH-1 Helicopters", *American Helicopter Society 57th Annual Forum*, American Helicopter Society, Washington D.C.

[14] Dempsey, P. J., 2000, "A Comparison of Vibration and Oil Debris Gear Damage Detection Methods Applied to Pitting Damage", NASA TM 2000-210371, NASA Glenn Research Center, Cleveland, OH.